Oil Whirl Effects on Rotor–Bearing System Identifications by Modal Testing

by
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1. Introduction

By modal testing the inherent dynamic characteristics of structures described by the modal parameters such as natural frequency, mode shape and damping ratio can be identified. The modal parameters obtained by modal testings are used to verify or to adjust the theoretical model, and to produce a mathematical model of components which is difficult to model in analytical form, and so on. Till now many works on the applications of modal testings to rotor systems were published, most applications were to identify the dynamic characteristics of fluid film bearings and seals.

Nordmann has applied the modal testing technique using impact technique to rotor systems to evaluate the stiffness and damping coefficients of bearings and seals [1,2]. Although the application of modal testing for parameter identification and diagnostics of rotor systems requires a special approach, Nordmann has applied the same theory which has been applied to non–rotating structures to rotor systems. Since all dynamic characteristics of rotors are closely related with rotor rotations, the directivity is very important in rotor dynamics. As a rotor starts rotating, two different kinds of modes known as forward and backward (precession) modes take place [3–4]. Modal characteristics associated with forward and backward modes are different from conventional modal characteristics of non–rotating structures. If the classical modal testing techniques is applied to rotors, the directivity of a mode of rotor systems are completely ignored. The forward and backward modes could not be distinguishable in frequency domain, and the frequency response characteristics of those two physically different modes are mixed up together over one–sided frequency region, resulting in heavy overlapping everywhere of the otherwise physically well separated modes. Recently new modal testing theories of rotor–bearing systems were published [5–7] to separate rotor vibrations into positive and negative frequency regions.

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Muszynska has also developed a perturbation equipment to give excitations rotating in only one direction to obtain the vibrations of rotors in separated (positive or negative) frequency regions[8–10]. Using the equipment she has been able to well separate frequency response functions into two frequency regions. Muszynska found that during modal testing using the perturbation equipment the vibrations due to oil whirl seriously affected the magnitude and phase of the frequency response functions. She also showed that the oil whirl effects did not appear in negative frequency region, whereas seriously appeard in positive frequency region. But in Nordmann's experiments using impact excitation technique the oil whirl effects did not appear in positive and negative frequency regions [1,2].

The equations of motion of rotor systems supported by fluid film bearings are often expressed as linearized form. With the linearized equations the resonances of rotor systems may well be evaluated. But the vibrations due to oil whirls can not be calculated because the vibrations are resulted from the nonlinear forces of fluid film bearings. Here using nonlinear fluid film forces oil whirl effects are disccused with modal testing methods.

2. Analysis of Fluid Film Bearing

A typical fluid film bearing is shown in Fig.1. The radial displacement of journal center, o', from the bearing center, o, is denoted as the eccentricity, e, of the journal, and when divided by the clearance, c, the eccentricity ratio, ϵ, may then take on values only from zero to unity. When the density is constant, the Reynolds' equation may be given as

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{R\partial\varphi}\left(\frac{h^3}{\mu}\frac{\partial p}{R\partial\varphi}\right) = 6U\frac{\partial p}{R\partial\varphi} + 12V \qquad (1)$$

The film thickness may be expressed as

$$h = c\,(1+\epsilon\cos\varphi) \qquad (2)$$

If the film thickness is "unwrapped", the velocities may be expressed by following components

$$U = R\Omega + c\dot{\epsilon}\sin\varphi - c\epsilon\dot{\vartheta}\cos\varphi$$
$$V = c\dot{\epsilon}\cos\varphi + c\epsilon\dot{\vartheta}\sin\varphi \qquad (3)$$

where U and V denote the velocities of journal surface at

φ in y znd z directions, respectively. Then Eq.(1) can be rewritten as, using Eqs.(2) and (3),

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{R\partial\varphi}\left(\frac{h^3}{\mu}\frac{\partial p}{R\partial\varphi}\right)=-6(\Omega-2\dot{\theta})c\epsilon\sin\varphi+12c\dot{\epsilon}\cos\varphi \quad (4)$$

Two basic approaches to the solution of Eq.(4) have been reported in the literature. If it is assumed that the fluid film bearing is very long, then it is possible to neglect the fluid flow and pressure gradients along the x-axis. On the other hand, if it is assumed that the bearing is relatively short, the appropriate approach is to neglect the flow in the φ direction in Fig.1 due to pressure gradients and arrive at

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) = -6(\Omega-2\dot{\theta})c\epsilon\sin\varphi +12\, c\dot{\epsilon}\cos\varphi \quad (5)$$

which is known as the governing equation for short bearing solution or Ocvirk model. In references [11–14] the short bearing assumption is very good for L/D ratios of 0.5 or less, or for L/D $\le$ 1.

With the assumption of short bearings Eq.(5) can be integrated directly and by applying the boundary conditions

$$p(\varphi,-\ell/2) = p(\varphi,\ell/2) = 0 \quad (6)$$

to evaluate the two constants of integration, then the following equation results

$$p = -3\frac{\mu}{c^2}\left(x^2-\frac{\ell^2}{4}\right)\left[\frac{\epsilon(\Omega-2\dot{\theta})\sin\varphi}{(1+\epsilon\cos\varphi)^3}-\frac{2\dot{\epsilon}\cos\varphi}{(1+\epsilon\cos\varphi)^3}\right] \quad (7)$$

When p = 0, $\tan\varphi = \dfrac{2\dot{\epsilon}}{\epsilon(\Omega-2\dot{\theta})}$ or

$$\left.\begin{matrix}\varphi_2\\ \varphi_1\end{matrix}\right\} = \tan^{-1}\left[\frac{2\dot{\epsilon}}{\epsilon(\Omega-2\dot{\theta})}\right] \quad (8)$$

where φ_1 is the angular location of the beginning of the film and φ_2 define the end of the film. From Eq.(8)

$$\varphi_2 = \varphi_1 + \pi \quad (9)$$

At $\varphi = \varphi_1$, $\partial p/\partial\varphi \ge 0$, and at $\varphi = \varphi_2$, $\partial p/\partial\varphi < 0$, then

$$\begin{aligned}\epsilon(\Omega-2\dot{\theta})\cos\varphi_1 + 2\dot{\epsilon}\sin\varphi_1 &= \frac{\epsilon(\Omega-2\dot{\theta})}{\cos\varphi_1} \ge 0\\ \epsilon(\Omega-2\dot{\theta})\cos\varphi_2 + 2\dot{\epsilon}\sin\varphi_2 &= \frac{\epsilon(\Omega-2\dot{\theta})}{\cos\varphi_2} < 0\end{aligned} \quad (10)$$

Therefore the following relations are given

$$\begin{aligned}&\cos\varphi_1 = \mathrm{sgn}(\Omega-2\dot{\theta})|\cos\varphi_1|,\ \cos\varphi_2= -\mathrm{sgn}(\Omega-2\dot{\theta})|\cos\varphi_2|\\ &\sin\varphi_1 = \mathrm{sgn}(\dot{\epsilon})|\sin\varphi_1|,\ \sin\varphi_2 = -\mathrm{sgn}(\dot{\epsilon})|\sin\varphi_2|\end{aligned} \quad (11)$$

where sgn denotes the sign of the value in parenthesis.

The resultant forces can be obtained from the integration of Eq.(7).

$$\begin{aligned}F_r &= \int_{\varphi_1}^{\varphi_2}\int_{-\frac{\ell}{2}}^{\frac{\ell}{2}} pR\cos\varphi\, dx\, d\varphi\\ F_t &= \int_{\varphi_1}^{\varphi_2}\int_{-\frac{\ell}{2}}^{\frac{\ell}{2}} pR\sin\varphi\, dx\, d\varphi\end{aligned} \quad (12)$$

where the subscripts r and t denote radial and tangential directions, respectively. In the integration of Eq.(12) $\varphi_1 = 0$, $\varphi_2 = \pi$ are assumed [12]. When $(\Omega-2\dot{\theta}) \ge 0$, resulting forces of Eq.(12) are given as

$$\begin{aligned}F_r &= -\mu R\ell\left(\frac{\ell}{c}\right)^2\left[(\Omega-2\dot{\theta})\frac{\epsilon^2}{(1-\epsilon^2)^2} + \dot{\epsilon}\,\frac{(1+2\epsilon^2)}{2(1-\epsilon^2)^{2.5}}\right]\\ F_t &= \mu R\ell\left(\frac{\ell}{c}\right)^2\left[(\Omega-2\dot{\theta})\frac{\pi\epsilon}{4(1-\epsilon^2)^{1.5}} + \dot{\epsilon}\,\frac{2\epsilon}{(1-\epsilon^2)^2}\right]\end{aligned} \quad (13)$$

When $(\Omega-2\dot{\theta}) < 0$, resulting forces of Eq.(12) are given as

$$\begin{aligned}F_r &= -\mu R\ell\left(\frac{\ell}{c}\right)^2\left[-(\Omega-2\dot{\theta})\frac{\epsilon^2}{(1-\epsilon^2)^2} + \dot{\epsilon}\,\frac{(1+2\epsilon^2)}{2(1-\epsilon^2)^{2.5}}\right]\\ F_t &= \mu R\ell\left(\frac{\ell}{c}\right)^2\left[(\Omega-2\dot{\theta})\frac{\pi\epsilon}{4(1-\epsilon^2)^{1.5}} - \dot{\epsilon}\,\frac{2\epsilon}{(1-\epsilon^2)^2}\right]\end{aligned} \quad (14)$$

The incremental fluid film forces may be expressed by linearized stiffness and damping coefficients in the small region near the steady state equilibrium position (y_o,z_o). The incremental forces are thus expressed by

$$\begin{aligned}\delta F_y &= -[\,k_{yy}y + k_{yz}z + c_{yy}\dot{y} + c_{yz}\dot{z}\,]\\ \delta F_z &= -[\,k_{zy}y + k_{zz}z + c_{zy}\dot{y} + c_{zz}\dot{z}\,]\end{aligned} \quad (15)$$

The detail expressions of Eq.(15) are given in [12,13].

3. Oil Whirl Effects on System Identifications by Modal Testing

Consider a simple rotor system shown in Fig.2. A rigid disk is mounted on the massless rigid shaft, and the shaft is supported by short fluid film bearings at both ends. The equations of motion are given as

$$\begin{aligned}m\,y &= F_{y_1} + F_{y_2} - mg + F_{yo}\\ m\,z &= F_{z_1} + F_{z_2} + F_{zo}\\ J_T y' + \Omega J_p \dot{z}' &= -F_{y_1}\ell_1 + F_{y_2}\ell_2\\ J_T z' - \Omega J_p \dot{y}' &= -F_{z_1}\ell_1 + F_{z_2}\ell_2\end{aligned} \quad (16)$$

where F_{y_1}, F_{y_2}, F_{z_1} and F_{z_2} are the nonlinear fluid film forces of the bearings as given in Eqs.(13) and (14), F_{yo} and F_{zo} external forces in y, z directions, respectively.

When R=L=1.5cm, c=0.005cm and μ=0.025kg/(m.sec) where R, L, c and μ are bearing radius, bearing length, bearing clearance and oil viscosity, respectively, the whirl speeds and logarithmic decrements using the linearized stiffness and damping coefficients are shown in Fig.3. From the rotational speed of 11500 rpm, the unstable whirl occurs. The unstable whirl frequency at the rotational speed of 11500 rpm is 5800 rpm, that is, almost the half of the rotational speed. When the rotational speed is 9000 rpm and the magnitude of excitation force is 10N, the forced response function using the nonlinear fluid film forces of Eqs.(13) and (14) is shown in Fig.4. As shown in Fig.4, there is a peak near the resonant frequency shown in Fig.3.(a). But oil whirl effects do not appear in the forced response function. When the magnitude of excitation force is 50 N, the forced response function is shown Fig.5. As shown in Fig.5 the ampltidue is greater than those of Fig.4, and oil whirl effects seriously appear near the half of rotational speed. When the backward excitation of 50 N acts, the forced response function is shown in Fig.6. As shown in Fig.6, the oil whirl effects do not appear in forced response function. In the Muszynska's experiments [8–10] the oil whirl effects did not appear in negative frequency region whereas the oil whirl effects seriously appeared in positive frequency region. When the uni–directional excitation in y direction acts, oil whirl effects do not appear in positive and negative frequency region as shown in Fig.7. In Nordmann's experiments using uni–directional impact excitations oil whirl effects did not appear in positive and negative frequency region [1,2]. When the rotational speeds are 5000 rpm, 8000 rpm, and 8500 rpm, the forced response function are shown in Figs.8–10, respectively. As shown in Fig.8, oil whirl effects do not appear when the eccentricity and the damping effects are great [10].

When rotational speeds are 5000 rpm, 9000 rpm and 12000 rpm, the whirl orbits at various excitation frequencies are shown in Figs.11–13, respectively. When the rotational speed is 9000 rpm, the large vibrations occur near the oil whirl resonance and system resonance. But at high excitation frequency as shown in Fig.12.(d), the vibration is small and stable. When the rotational speed is 12000 rpm, unstable whirls occur regardless of excitation frequency as shown in Fig.13. As shown in Fig.3.(a) beyond the rotational speed of 11500 rpm unstable vibrations occur.

The most applications of modal testings to rotor systems are to identify the dynamic characteristics of fluid film bearings and seals [1–2,8–10]. During modal testings oil whirl effects on forced responses are seriously dependent on excitation methods. When uni–directional excitation technique is used, oil whirl effects on forced responses can be minimized. In other words the linearized dynamic coefficients of fluid film bearings and seals can be estimated more accurately with the uni–directional excitation technique than with circular rotating excitation technique. With the frequency response functions obtained by uni–directional excitation technique the frequency response functions in which the vibrations of rotors are separated into positive and negative frequency regions can be obtained by the modal analysis theroies of [5–7]. But with the circular rotating excitation oil whirl effects can be well investigated [8–10].

4. Conclusions

Oil whirl effects on system identification during modal testings are discussed. When the forward rotating excitations act, the oil whirl effects seriously appear, But when the backward rotating and uni–directional excitations act, and the magnitude of forward excitation is small, oil whirl effects do not appear in forced response function. The results of simulation of oil whirl effects during modal testing are well coincident with those of experiments. With the uni–directional excitation technique the linearized dynamic coefficients of fluid film bearings and seals can be estimated more accurately than with the circular rotating excitation technique. But with the circular excitation technique oil whirl effects can be well investigated

Acknowledgements

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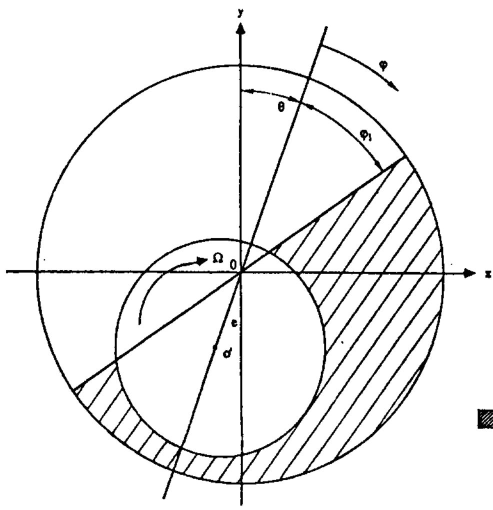


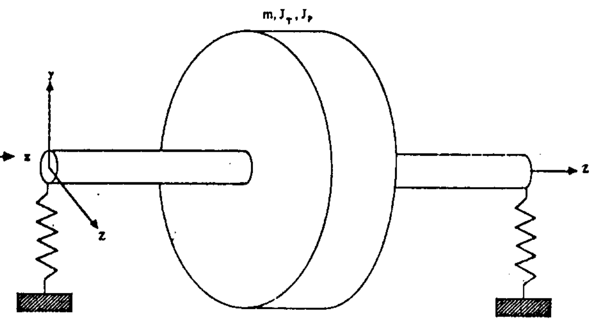
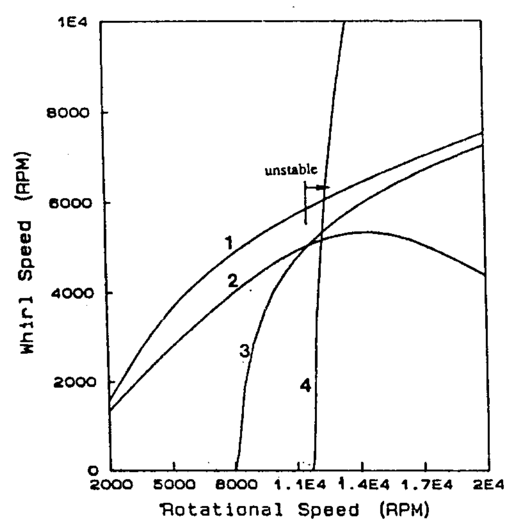
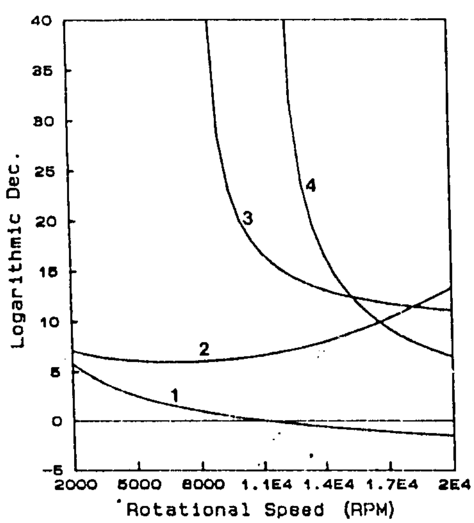


Figure 2 Simple Rotor System

Figure 1 Cross–Section of a Rigid Rotor System showing Fluid–Film



(a) Whirl Speed



(b) Logarithmic Decrement

Figure 3 Whirl Speeds and Logarithmic Decrements of Simple Rotor System Supported by Fluid Film Bearings ($R=L=1.5$cm, $c=0.005$cm, $\mu=0.025$ kg/(m.sec))

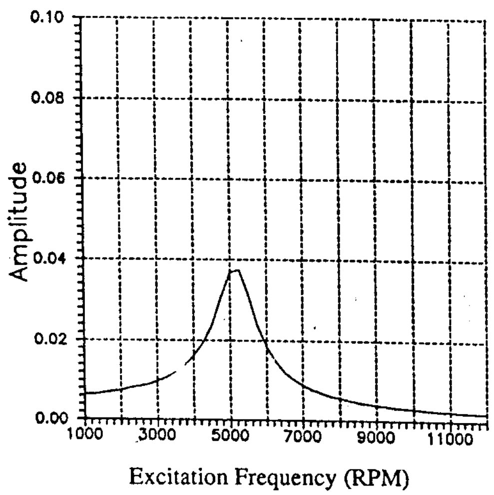


Figure 4 Forced Response Function when Forward Excitations act (F=10N, Ω=9000rpm)

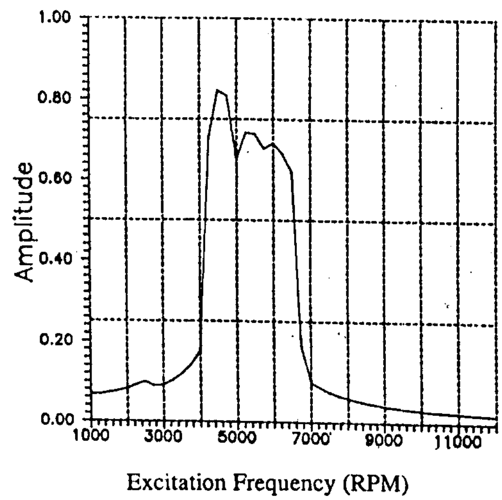


Figure 5 Forced Response Function when Forward Excitations act (F=50N, Ω=9000rpm)

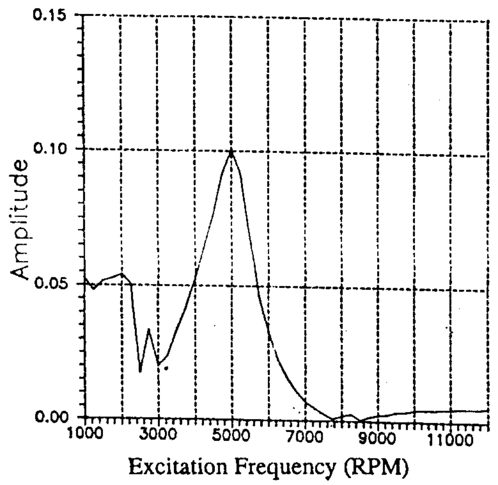


Figure 6 Forced Response Function when Backward Excitations act (F=50N, Ω=9000rpm)

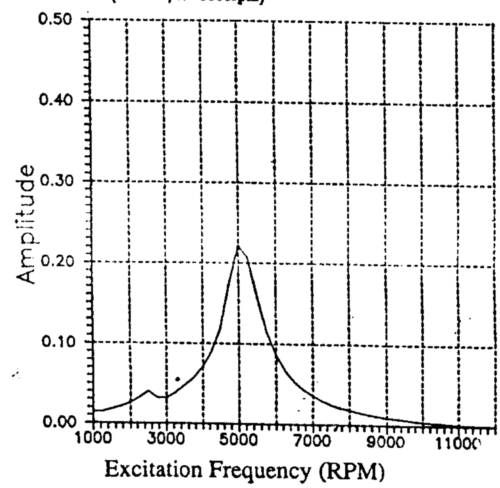


Figure 7 Forced Response Function when Uni-directional Excitations act (F=50N, Ω=9000rpm)

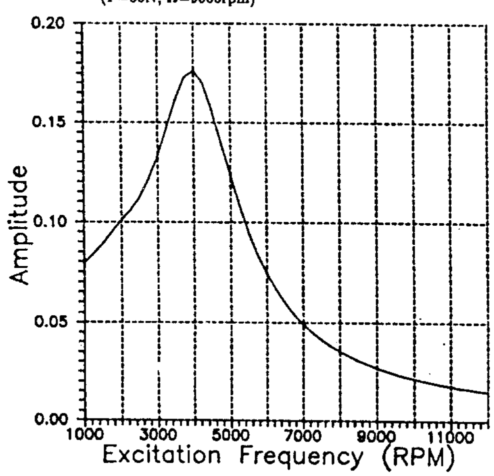


Figure 8 Forced Response Function when Forward Excitations act (F=50N, Ω=5000rpm)

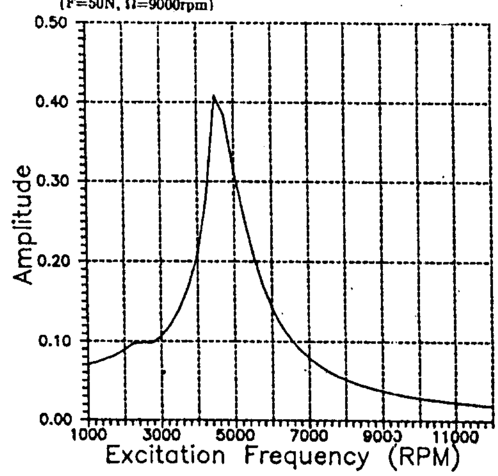
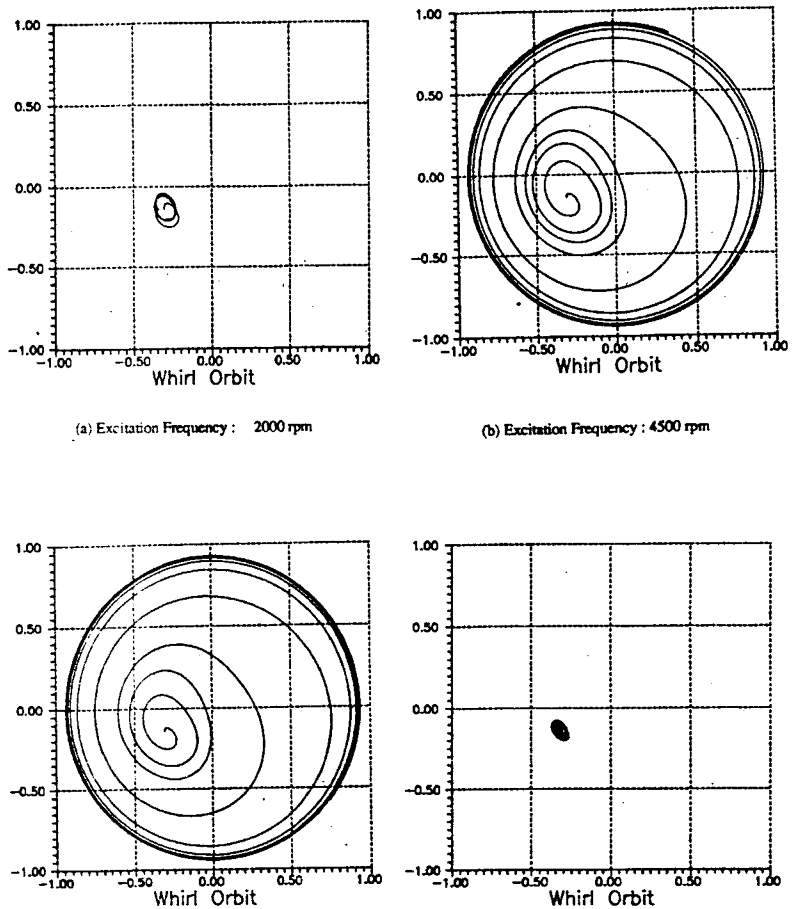


Figure 9 Forced Response Function when Forward Excitations act (F=50N, Ω=8000rpm)



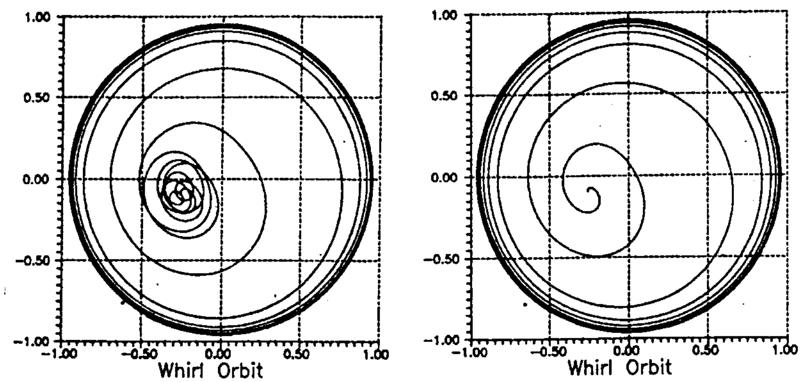
(a) Excitation Frequency : 2000 rpm

(b) Excitation Frequency : 4500 rpm

(c) Excitation Frequency : 5100 rpm

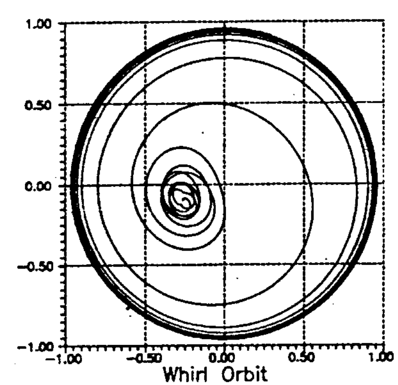
(d) Excitation Frequency : 10000 rpm

Figure 12 Whirl Orbits at the Rotational Speed of 9000 rpm



(a) Excitation Frequency : 2000 rpm

(b) Excitation Frequency : 5000 rpm



(c) Excitation Frequency : 10000 rpm

Figure 13 Whirl Orbits at the Rotational Speed of 12000 rpm

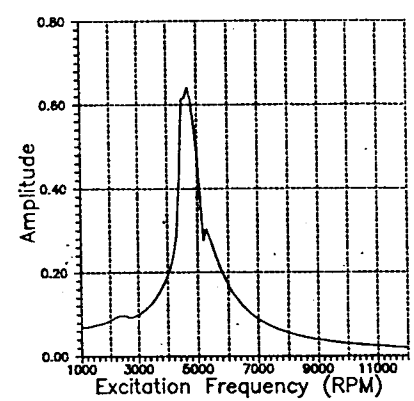


Figure 10 Forced Response Function when Forward Excitations act (F=50N, Ω=8500rpm)